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WITNESS my hand this
Twenty-seventh day of September 2000

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Patents Act 1990

PROVISIONAL SPECIFICATION

FOR THE INVENTION ENTITLED:

"A Rotary Blood Pump with Hydrodynamically Suspended Impeller"

Applicants: Cortronix Pty Ltd and University of Technology, Sydney

The invention is described in the following statement:

A ROTARY BLOOD PUMP WITH HYDRODYNAMICALLY SUSPENDED IMPELLER

Field of the Invention

This invention relates to rotary blood pumps for artificial hearts or ventricular assist devices and, in particular, discloses a design for a seal-less shaft-less centrifugal pump featuring open impeller blades with the blade edges used as hydrodynamic thrust bearings and with electromagnetic torque provided by the interaction between magnets embedded in the blades and a rotating current pattern generated in wires fixed relative to the pump housing.

Background Art

This invention relates to the art of continuous flow rotary pumps and, in particular, to electrically driven pumps suitable for use as an artificial heart or ventricular assist device. For permanent implantation in a human patient, such pumps should ideally have the following characteristics: no leakage of fluids into or from the bloodstream; no wearing parts; minimum residence time of blood in pump to avoid thrombosis (clotting); minimum shear stress on blood to avoid blood cell damage such as haemolysis; maximum efficiency to maximise battery duration and minimise blood heating; and absolute reliability.

Several of these characteristics are very difficult to meet in a conventional pump configuration including a seal, i.e. with an impeller mounted on a shaft which penetrates a wall of the pumping cavity, as exemplified by the blood pumps referred to in U.S. Pat. No. 3,957,389 to Rafferty et al., U.S. Pat. No. 4,625,712 to Wampler, and U.S. Pat. No. 5,275,580 to Yamazaki. Two main disadvantages of such pumps are firstly that the seal needed on the shaft may leak, especially after wear, and secondly that the rotor of the motor providing the shaft torque remains to be supported, with mechanical bearings such as ball-bearings precluded due to wear. Some designs, such as U.S. Pat. No. 4,625,712 to Wampler and U.S. Pat. No. 4,908,012 to Moise et al., have overcome these problems simultaneously by combining the seal and the bearing into one hydrodynamic bearing, but in order to prevent long residence times they have had to introduce means to continuously supply a

blood-compatible bearing purge fluid via a percutaneous tube. In seal-less designs, blood is permitted to flow through the gap in the motor, which is usually of the brushless DC type, i.e. comprising a rotor including permanent magnets and a stator in which an electric current pattern is made to rotate synchronously with the rotor. Such designs can be classified according to the means by which the rotor is suspended: contact bearings, magnetic bearings or hydrodynamic bearings, though some designs use two of these means.

Contact or pivot bearings, as exemplified by U.S. Pat. No. 5527159 to Bozeman et al. and U.S. Pat. No. 5399074 to Nosé et al., have potential problems due to wear, and cause very high localised heating and shearing of the blood, which can cause deposition and denaturation of plasma proteins, with the risk of embolisation and bearing seizure..

Magnetic bearings, as exemplified by U.S. Pat. No. 5,350,283 to Nakazeki et al., U.S. Pat. No. 5,326,344 to Bramm et al. and U.S. Pat. No. 4,779,614 to Moise et al., offer contactless suspension, but require rotor position measurement and active control of electric current for stabilisation of the position in at least one direction, according to Earnshaw's theorem. Position measurement and feedback control introduce significant complexity, increasing the failure risk. Power use by the control current implies reduced overall efficiency. Furthermore, size, mass, component count and cost are all increased.

US Pat. No. 5,507,629 to Jarvik claims to have found a configuration circumventing Earnshaw's Theorem and thus requiring only passive magnetic bearings, but this is doubtful and contact axial bearings are included in any case. Similarly, passive radial magnetic bearings and a pivot point are employed in U.S. Pat. No. 5,443,503 to Yamane.

Prior to the present invention, pumps employing hydrodynamic suspension, such as US Pat. No. 5,211,546 to Isaacson et al. and US Pat. No. 5,324,177 to Golding et al., have used journal bearings, in which radial suspension is provided by the fluid motion between two cylinders in relative rotation, an inner cylinder lying within and slightly off axis to a slightly

larger diameter outer cylinder. Axial suspension is provided magnetically in US Pat. No. 5,324,177 and by either a contact bearing or a hydrodynamic thrust bearing in US Pat. No. 5,211,546.

A purging flow is needed through the journal bearing, a high shear region, in order to remove dissipated heat and to prevent long fluid residence time. It would be inefficient to pass all the fluid through the bearing gap, of small cross-sectional area, as this would demand an excessive pressure drop across the bearing. Instead a leakage path is generally provided from the high pressure pump outlet, through the bearings and back to the low pressure pump inlet, implying a small reduction in outflow and pumping efficiency. US Pat. No. 5,324,177 provides a combination of additional means to increase the purge flow, namely helical grooves in one of the bearing surfaces, and a small additional set of impellers.

US Pat. No. 5,211,546 provides 10 embodiments with various locations of cylindrical bearing surfaces. One of these embodiments, the third, features a single journal bearing and a contact axial bearing.

Embodiments of the present invention offer a relatively low cost and/or relatively low complexity means of suspending the rotor of a seal-less blood pump, thereby overcoming or ameliorating the problems of existing devices mentioned above.

Summary of the Invention

According to one aspect of the present invention, there is disclosed a rotary blood pump with impeller suspended hydrodynamically by thrust forces generated on the edges of the impeller blades. The blade edges are shaped such that the gap at the leading edge is greater than at the trailing edge and thus the fluid which is drawn through the gap experiences a wedge shaped restriction which generates a thrust, as described in Reynold's theory of lubrication.

In the preferred embodiment of the invention, the pump is of centrifugal type with impeller blades open on both the front and back faces of the housing. The front face of the housing is made conical, in order that the thrust perpendicular to it has a radial component, which provides a radial restoring force to

a radial displacement of the impeller axis. Similarly, an axial displacement toward either the front or the back face increases the thrust from that face and reduces the thrust from the other face. Thus the sum of the forces on the impeller due to inertia (within limits), gravity and any bulk radial or axial hydrodynamic force on the impeller can be countered by a restoring force from the thrust bearings after a small displacement of the impeller within the housing.

In the preferred embodiment, the impeller driving torque derives from the magnetic interaction between permanent magnets within the blades of the impeller and oscillating currents in windings encapsulated in the pump housing.

In a second embodiment of the invention, the principle is applied in a pump of axial type. Within a uniform cylindrical section of the pump housing, tapered blade edges form a radial hydrodynamic bearing. If the pump housing is made with reducing radius at the two ends, then the end hydrodynamic thrust forces have an axial component which can provide the axial bearing. Alternatively, magnetic forces or other means can provide the axial bearing.

In a further broad form of the invention there is provided a rotary blood pump having an impeller suspended hydrodynamically by thrust forces generated by the impeller during movement in use of the impeller.

Preferably said thrust forces are generated by blades of said impeller.

More preferably said thrust forces are generated by edges of said blades of said impeller.

Preferably said edges of said blades are tapered.

In an alternative preferred form said pump is of axial type.

Preferably within a uniform cylindrical section of the pump housing, tapered blade edges form a radial hydrodynamic bearing.

Preferably the pump housing is made with reducing radius at the two ends, and wherein the end hydrodynamic thrust forces have an axial component which can provide the axial bearing.

Preferably magnetic forces or other means can provide the axial bearing.

In a further broad form invention there is provided a rotary blood pump having a housing within which an impeller acts by rotation about an axis to cause a pressure differential between an inlet side of a housing of said pump and an outlet side of the housing of said pump; said impeller suspended hydrodynamically by thrust forces generated by the impeller during movement in use of the impeller.

The Drawings

The present invention is now described, with reference to the accompanying drawings, wherein:

Fig. 1 is a longitudinal cross-sectional view of the preferred embodiment of the invention;

Fig. 2 is a cross-sectional view taken generally along the lines 2(2 of Fig. 1;

Fig. 3A is a cross-sectional view of an impeller blade taken generally along the lines 3A(3A of Fig. 2;

Fig. 3B is an enlargement of part of Fig. 3A;

Fig. 3C is an alternative impeller blade shape;

Fig. 4A-C illustrate various alternative locations of magnet material within a blade;

Fig. 5 is a left-hand end view of a possible winding geometry taken generally along the lines 5(5 of Fig. 1;

Fig. 6 is a diagrammatic cross-sectional view of an alternative embodiment of the invention as an axial pump.

Detailed Description

The preferred embodiment of the invention is the centrifugal pump 1, as depicted in Figs. 1 and 2, intended for implantation into a human, in which case the fluid referred to below is blood. The pump housing 2, can be fabricated in two parts, a front part 3 and a back part 4, with a smooth join, for example at 5 in Fig. 1. The pump has an axial inlet 6 and a tangential outlet 7. The rotating part is of very simple form, comprising only blades 8 and a support cone 9 to hold those blades fixed relative to each other. The blades may be curved as depicted in Fig. 2, or straight, in which case they can be either radial or tilted, i.e. at an angle to the radius. This rotating part will hereafter be called the impeller, but it also serves as a bearing component and as the rotor of the motor. Note that the

impeller has no shaft and that the fluid enters the impeller from its axis. Some of the fluid passes in front of the support cone and some behind it, so that the pump can be considered of two-sided open type, as compared to conventional open centrifugal pumps, which are only open on the front side. Approximate dimensions found adequate for the pump to perform as a ventricular assist device, when operating at speeds in the range 2,000 rpm to 4,000 rpm, are outer blade diameter 40 mm, outer housing average diameter 60 mm, and housing axial length 40 mm.

As the blade edges move around the housing, some of the fluid passes through the gaps, much exaggerated in Figs. 1 and 3, between the blades and the housing front face 10 and housing back face 11. In all open centrifugal pumps, the gaps are made small because this leakage flow lowers the pump hydrodynamic efficiency. In the pump disclosed in this invention, the gaps are made slightly smaller than is conventional in order that the leakage flow can be utilised to create a hydrodynamic bearing. For the hydrodynamic forces to be sufficient, the blades must also be tapered as depicted in Figs. 3A and 3B, so that the gap is larger at the leading edge of the blade than at the trailing edge. The fluid which passes through the gap thus experiences a wedge shaped restriction which generates a thrust, as described in Reynold's theory of lubrication (see, for example, "Modern Fluid Dynamics, Vol. 1 Incompressible Flow", by N. Curle and H.J. Davies, Van Nostrand, 1968). The thrust is proportional to the square of the blade thickness at the edge, and thus thick blades are favoured, since if the proportion of the pump cavity filled by blades is constant, then the net thrust force will be inversely proportional to the number of blades. However, the blades edges could be made to extend as tails from thin blades as depicted in Fig. 3C.

For manufacturing simplicity, the housing front face 10 can be made conical, with an angle of around 45(so that it provides both axial and radial hydrodynamic forces. The housing back face 11 should include a roughly conical extension 12 pointing into the pump cavity, to eliminate the flow stagnation point on the axis of the back housing. In the preferred embodiment, for

manufacturing simplicity and for uniformity in the flow axial direction, the housing back face 11 is made flat over the bearing surfaces, i.e. under the blade edges. With this the case, a slacker tolerance on the alignment between the axes of the front and back halves of the housing is permissible. An alternative would be to make the back face conical at the bearing surfaces, with taper in the opposite direction to the front face, so that the hydrodynamic forces from the back face would also have radial components. Tighter tolerance on the axes alignment would then be required, and some of the flow would have to undergo a reversal in its axial direction. Again a roughly conical extension like 12 would be needed. There may be some advantage in making the housing surfaces and blade edges non-straight, with varying tangent angle, but this would impose greater manufacturing complexity.

There are several options for the shape of the taper, but in the preferred embodiment the amount of material removed simply varies linearly or approximately linearly across the blade. For the back face, the resulting blade edges are then planes at a slight inclination to the back face. For the front face, the initial blade edges are curved and the taper only removes a relatively small amount of material so they still appear curved. Alternative taper shapes include a step in the blade edge, though the corner in that step would represent a stagnation line posing a thrombosis risk.

For a given minimum gap, at the trailing blade edge, the hydrodynamic force is maximal if the gap at the leading edge is approximately double that at the trailing edge. Thus the taper, which equals the leading edge gap minus the trailing edge gap, should be chosen to match a nominal minimum gap, once the impeller has shifted towards that edge. Dimensions which have been found to give adequate thrust forces are a taper of around 0.05 mm for a nominal minimum gap of around 0.05 mm, and an average circumferential blade edge thickness of around 5 mm for 4 blades. For the front face, the taper is measured within the plane perpendicular to the axis. The axial length of the housing between the front and back faces at any position should then be made about 0.2 mm greater than the axial length of the

blade, when it is coaxial with the housing, so that the minimum gaps are both about 0.1 mm axially when the impeller is centrally positioned. Then, for example, if the impeller shifts axially by 0.05 mm, the minimum gaps will be 0.05 mm at one face and 0.15 mm at the other face. The thrust increases with decreasing gap and would be much larger from the 0.05 mm gap than from the 0.15 mm gap, to be precise about 14 times larger for the above dimensions. Thus there is a net restoring force away from the smaller gap.

Similarly, for radial shifts of the impeller the radial component of the thrust from the smaller gap on the conical housing front face would offer the required restoring radial force. The axial component of that force and its torque on the impeller would have to be balanced by an axial force and torque from the housing back face, and so the impeller would also have to shift axially and tilt its axis to be no longer parallel with the housing axis. Thus as the person moves pump is accelerated by external forces, the impeller will continually shift its position and alignment, varying the gaps in such a way that the total force and torque on the impeller match that demanded by inertia. The gaps are so small, however, that the variation in hydrodynamic efficiency will be small, and the pumping action of the blades will be approximately the same as when the impeller is centrally located.

While smaller gaps imply greater hydrodynamic efficiency and greater bearing thrust forces, smaller gaps also demand tighter manufacturing tolerances, increase frictional drag on the impeller, and impose greater shear stress on the fluid. Taking these points in turn, for the above 0.05 mm tapers and gaps, tolerances of around (0.015 mm are needed, which imposes some cost penalty but is achievable. A tighter tolerance would be difficult, especially if the housing is made of a plastic, given the changes in dimension caused by temperature and possible absorption of fluid.water absorptivity of plastic. The frictional drag for the above gaps produces negligible torque compared to the typical motor torque. Finally, to estimate the shear stress, consider a rotation speed of 3,000 rpm and a typical radius of 15 mm, at which the blade speed is 4.7 ms-1

and the average velocity shear for an average gap of 0.075 mm is 6.2 (104 s-1. For blood of dynamic viscosity 3.5 (10-3 kgm-1s-1, the average shear stress would be 220 Nm-2. Other prototype centrifugal blood pumps with closed blades have found that slightly larger gaps, 0.15 mm, are acceptable for haemolysis. A major advantage of the open blades of the present invention is that a fluid element that does pass through a blade edge gap will have very short residence time in that gap, around 2 (10-3 s, and the fluid element will most likely be swept though the pump without passing another blade edge. In contrast, in closed or open-closed centrifugal pumps, the residence time of a fluid element passing between the housing and a closed impeller face is determined by the purge rate and could be seconds.

To minimise the net force required of the hydrodynamic bearings, the net axial and radial hydrodynamic forces on the impeller from the bulk fluid flow should be minimised, where "bulk" here means other than from the bearing thrust surfaces.

One method of minimising the bulk radial hyrodynamic force is to use straight radial blades so that pressure acting on the blade sides has virtually no radial component. The radial force on the impeller depends critically on the shape of the output flow collector or volute 13. The shape should be designed to minimise the radial impeller force over the desired range of pump speeds, without excessively lowering the pump efficiency. The optimal shape will have a roughly helical perimeter between the "tongue" and outlet. The radial force can also be reduced by the introduction of an internal division in the volute to create a second output flow collector passage, with tongue approximately diametrically opposite to the tongue of the first passage.

In regard to the bulk axial hyrodynamic axial force, if the blade cross-section is made uniform in the axial direction, apart from the conical front edge, then the pressure acting on the blade surface (excluding the bearing edges) will have no axial component. This also simplifies the blade manufacture. The blade support cone 9 must then be shaped to minimise axial thrust on the impeller and minimise disturbance to the flow

over the range of speeds, while maintaining sufficient strength to prevent relative blade movement. The key design parameter affecting the axial force is the angle of the cone. The cone is drawn in Fig. 1 as having the same internal diameter as the blades, which may aid manufacture. However, the cone could be made with larger or smaller internal diameter to the blades. There may be advantage in using a non-axisymmetric support "cone", e.g. with larger radius on the trailing surface of a blade than the radius at the leading surface of the next blade. If the blades are made with non-uniform cross-section to increase hydrodynamic efficiency, then any bulk hydrodynamic axial force on them can be balanced by shaping the support cone to produce an opposite bulk hydrodynamic axial force on it. Careful design of the entire pump, employing computational fluid dynamics, is necessary to determine the optimal shapes of the blades, the volute, the support cone and the housing, in order to maximise hydrodynamic efficiency while keeping the bulk fluid hydrodynamic forces, shear and residence times low. All edges and the joins between the blades and the support cone should be smoothed.

The means of providing the driving torque on the impeller of the preferred embodiment of the invention is to encapsulate permanent magnets 14 in the blades 8 of the impeller and to drive them with the rotating magnetic field pattern from oscillating currents in windings 15 and 16, fixed relative to the housing. Magnets of high remanence such as sintered rare-earth magnets should be used to maximise motor efficiency. The magnets should be aligned axially or approximately axially, with alternating polarity for adjacent blades. Thus there must be an even number of blades. Since low blade number is preferred for the bearing force, and since 2 blades would not have sufficient bearing stiffness to rotation about an axis through the blades and perpendicular to the pump housing (unless the blades are very curved), 4 blades is are recommended.

Some possible options for locating the magnets within the blades are suggested in Fig. 4. The ideal, depicted Fig. 4A, is for the blade to be made of magnet apart from a biocompatible

shell or coating to prevent fluid corroding the magnets and to prevent magnet material (which is toxic for rare-earth magnets) entering the blood stream. The coating should also be sufficiently durable especially at blade corners to withstand rubbing during start-up or during inadvertent bearing touch down. A possible impeller manufacturing method is to die-press the entire impeller, blades and support cone, as a single axially aligned magnet. The die-pressing is much simplified if near axially uniform blades are used (blades with an overhang such as in Fig. 3C are precluded). During pressing, the crushed rare-earth particles must be aligned in an axial magnetic field. This method of die-pressing with parallel alignment direction is cheaper for rare-earth magnets, though produces slightly lower remanence magnets. The tolerance in die-pressing is poor, and grinding of the tapered blade edges is required. Then the magnet impeller can be coated, for example by physical vapour deposition, of titanium nitride for example, or by chemical vapour deposition, of with a thin diamond coating for example by chemical vapour deposition or a teflon coating. Finally, to create the alternating blade polarity the impeller must be placed in a special pulse magnetisation fixture, with an individual coil surrounding each blade. The support cone acquires some magnetisation near the blades, with negligible influence.

Alternative magnet locations are sketched in Fig. 4B and Fig. 4C in which quadrilateral or circular cross-section magnets are inserted into the blades. Sealing and smoothing of the blade edges over the insertion holes would then be required to reinstate the taper.

All edges in the pump should be radiused and surfaces smoothed to avoid thrombosis.

The windings 15 and 16 of the preferred embodiment are slotless or air-gap windings, following the blade curvature, with the same pole number as the impeller, namely 4 poles in the preferred embodiment. An ferromagnetic iron yoke of conical form 17 for the front winding and an iron ferromagnetic yoke of annular form 18 for the back winding may be placed on the outside of the windings to increase the magnetic flux densities

and hence increase motor efficiency. The winding thicknesses should be designed for maximum motor efficiency, with the sum of their axial thicknesses somewhat less than but comparable to the magnet axial length. The yokes can be made of solid ferromagnetic material such as iron, or to reduce "iron" losses, the yokes can be laminated, for example by helically winding thin strip, or can be made of iron/powder epoxy composite. or can be helically wound to reduce iron losses. The yokes should be positioned such that there is zero net axial magnetic force on the impeller when it is positioned centrally in the housing. The magnetic force is unstable and increases linearly with axial displacement of the impeller away from the central position, with the gradient being called the positive stiffness of the magnetic force. This unstable magnetic force must be countered by the hydrodynamic bearings, and so the stiffness should be made as small as possible. Choosing the yoke thickness such that the flux density is at the saturation level reduces the stiffness and gives minimum mass. An alternative would be to have no iron yokes, completely eliminating the unstable axial magnetic force, but the efficiency of such designs would be lower and the magnetic flux density in the immediate vicinity of the pump may violate safety standards and produce some tissue heating. In any case, the stiffness is acceptably small for slotless windings with the yokes present. Another alternative would be to insert the windings in slots in laminated iron stators which would increase motor efficiency and enable less magnet material and potentially lighter impeller blades. However, the unstable magnetic forces would be significant for such slotted motors. Also, the necessity for fat blades to generate the required bearing forces allows room for large magnets, and so slotless windings are chosen in the preferred embodiment.

Fig. 5 depicts one suitable topology for the front face winding 15. The back face winding 16 looks similar from the back end of the motor, except the hole on the axis is smaller. Each winding has three phases, A, B and C, and two coils connected in series or parallel per phase. Each coil comprises a number of turns of an insulated conductor such as copper, with the number of turns

chosen to suit the desired voltage. The conductor may need to be stranded to reduce eddy losses. The winding construction can be simplified by laying the coils around pins protruding from a temporary conical former, the pins shown as dots in 2 rings of 6 pins each in Fig. 5. The coils are labelled alphabetically in the order in which they would be layed, coils a and d for phase A, b and e for phase B, and c and f for phase C. Instead of or as well as pins, the coil locations could be defined by thin curved fins, running between the pins in Fig. 5, along the boundary between the coils.

The winding connection of the preferred embodiment is for three wires, one wire per phase, to connect a sensorless electronic controller to winding 15, three wires to pass between windings 15 and 16, and for a neutral point termination of the wires within winding 16. A neutral lead, N in Fig. 5, between the controller and the neutral point is optional. A standard sensorless controller can be used, in which two out of six semiconducting switches in a three phase bridge are turned on at any one time, with the switching synchronised with the impeller position via the back-emf in the unenergised phase. Alternatively, because of the relatively small fraction of the impeller cross-section occupied by magnets, it may be slightly more efficient to only activate one of the three phases at a time, and to return the current by a fourth wire from the winding 16 neutral point back to the controller. The provision of the neutral lead also enables redundancy to be built into the motor and controller, so that if any one of the three phases fails in either the motor or controller, then the other two phases can still provide a rotating magnetic field sufficient to drive the pump. Careful attention must be paid to ensure that the integrity of all leads and connections is failsafe.

In the preferred embodiment, the two housing components 3 and 4 are made by injection moulding from non-conducting plastic materials such as Lexan polycarbonate plastic or ceramics, and the windings and yokes are encapsulated within the housing during fabrication moulding. In this way, the separation between the winding and the magnets is minimised, increasing

the motor efficiency, and the housing is thick, increasing its mechanical stiffness. Alternatively, the windings can be positioned outside the housing, of thickness at least around 2 mm for sufficient stiffness.

If the housing material plastic is hygroscopic or if the windings are outside the housing, it may be necessary to first enclose the windings and yoke in a very thin impermeable shell. Ideally the shell should be non-conducting (such as ceramic or plastic), but titanium of around 0.1 mm to 0.2 mm thickness would give sufficiently low eddy losses. Encapsulation within such a shell would be needed to prevent winding movement.

By keeping the windings separate for the front and back faces, the windings can be moulded into the front and back housing parts. Alternatively, for the case of windings not moulded into the housings, it would be possible to wind the coils onto the assembled housing, passing the coils from the front face to the back face over the volute 13. This may slightly reduce "end-winding" lengths and hence increase motor efficiency.

The combining of the motor and bearing components into the impeller in the preferred embodiment provides several key advantages. The rotor consequently has very simple form, with the only cost of the bearing being tight manufacturing tolerances. The rotor mass is very low, minimising the bearing force needed to overcome weight. Also, with the bearings and the motor in the same region of the rotor, the bearings forces are smaller than if they had to provide a torque to support magnets at an extremity of the rotor.

A disadvantage of the combination of functions in the impeller is that its design is a coupled problem. The optimisation should ideally link the fluid dynamics, magnetics and bearing thrust calculations. In reality, the blade thickness can be first roughly sized to give adequate motor efficiency and sufficient bearing forces with a safety margin. Fortuitously, both requirements are met for 4 blades of approximate average circumferential thickness 5 mm. The housing, blade, and support cone shapes can then be designed using computational fluid dynamics, maintaining the above minimum average blade thickness. Finally the motor stator, i.e. winding and yoke, can

be optimised for maximum motor efficiency.

Fig. 6 depicts an alternative embodiment of the invention as an axial pump. The pump housing is made of two parts, a front part 19 and a back part 20, joined for example at 21. The pump has an axial inlet 22 and axial outlet 23. The impeller comprises only blades 24 mounted on a support cylinder 25 of reducing radius at each end. The essential feature of the design is that the blade edges are tapered to generate hydrodynamic thrust forces which suspend the impeller. These forces could be used for radial suspension alone from the straight section 26 of the housing, with some alternative means used for axial suspension, such as stable axial magnetic forces or a conventional tapered-land type hydrodynamic thrust bearing. Fig. 6 proposes a design which uses the tapered blade edges to also provide an axial hydrodynamic bearing. The housing is made with a reducing radius at its ends to form a front face 27 and a back face 28 from which the axial thrusts can suspend the motor axially. Magnets are embedded in the blades with blades having alternating polarity and 4 blades being recommended. Iron in the outer radius of the support cylinder 25 can be used to increase the magnet flux density. Alternatively, the magnets could be housed in the support cylinder and iron could be used in the blades. A slotless helical winding 29 is recommended, with outward bending end-windings 30 at one end to enable insertion of the impeller and inward bending windings 31 at the other end to enable insertion of the winding onto a cylindrical magnetic yoke 32. The winding can be encapsulated in the back housing part 20.

The foregoing describes the principles of the present invention, and modifications, obvious to those skilled in the art, can be made thereto without departing from the scope of the invention.

CLAIMS

1. A rotary blood pump having an impeller suspended hydrodynamically by thrust forces generated by the impeller during movement in use of the impeller.

2. The blood pump of claim 1 wherein said thrust forces are generated by blades of said impeller.

3. The blood pump of claim 2 wherein said thrust forces are generated by edges of said blades of said impeller.

4. The blood pump of claim 3 wherein said edges of said blades are tapered.

5. The blood pump of any one of claims 2 to 4 wherein said blades edges are shaped such that the gap at the leading edge is greater than at the trailing edge and thus the fluid which is drawn through the gap experiences a wedge shaped restriction which generates a thrust.

6. The blood pump of any previous claim wherein the pump is of centrifugal type with impeller blades open on both front and back faces of the pump housing.

7. The blood pump of claim 6 wherein the front face of the housing is made conical, in order that the thrust perpendicular to it has a radial component, which provides a radial restoring force to a radial displacement of the impeller axis.

8. The blood pump of any previous claim wherein the driving torque of said impeller derives from the magnetic interaction between permanent magnets within the blades of the impeller and oscillating currents in windings encapsulated in the pump housing.

9. The rotary blood pump of any previous claim wherein said pump is of axial type.

10. The rotary blood pump of claim 9 wherein within a uniform cylindrical section of the pump housing, tapered blade edges form a radial hydrodynamic bearing.

11. The rotary blood pump of claim 9 or claim 10 wherein the pump housing is made with reducing radius at the two ends, and wherein the end hydrodynamic thrust forces have an axial component which can provide the axial bearing.

12. The rotary blood pump of claim 9 or claim 10 wherein magnetic forces or other means can provide the axial bearing.

13. A rotary blood pump having a housing within which an impeller acts by rotation about an axis to cause a pressure differential between an inlet side of a housing of said pump and an outlet side of the housing of said pump; said impeller suspended hydrodynamically by thrust forces generated by the impeller during movement in use of the impeller.

DATED: 5 September, 1997

CARTER SMITH & BEADLE

Patent Attorneys for the Applicant:

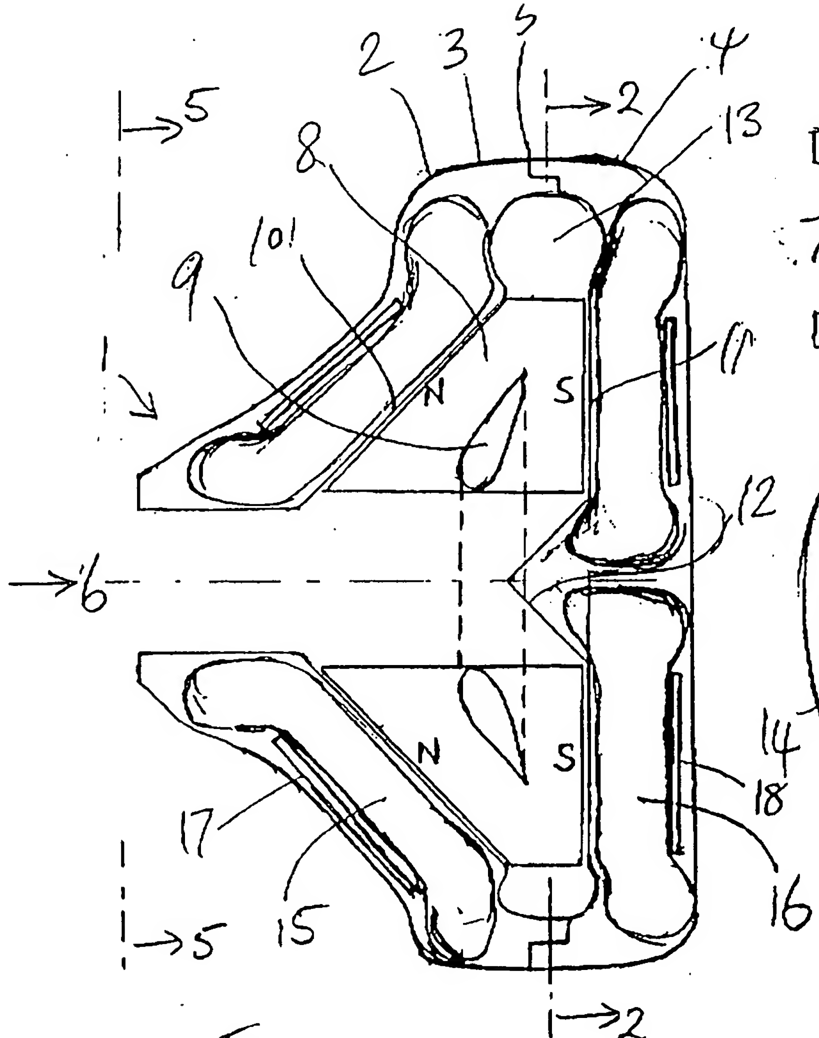

FIG. 1

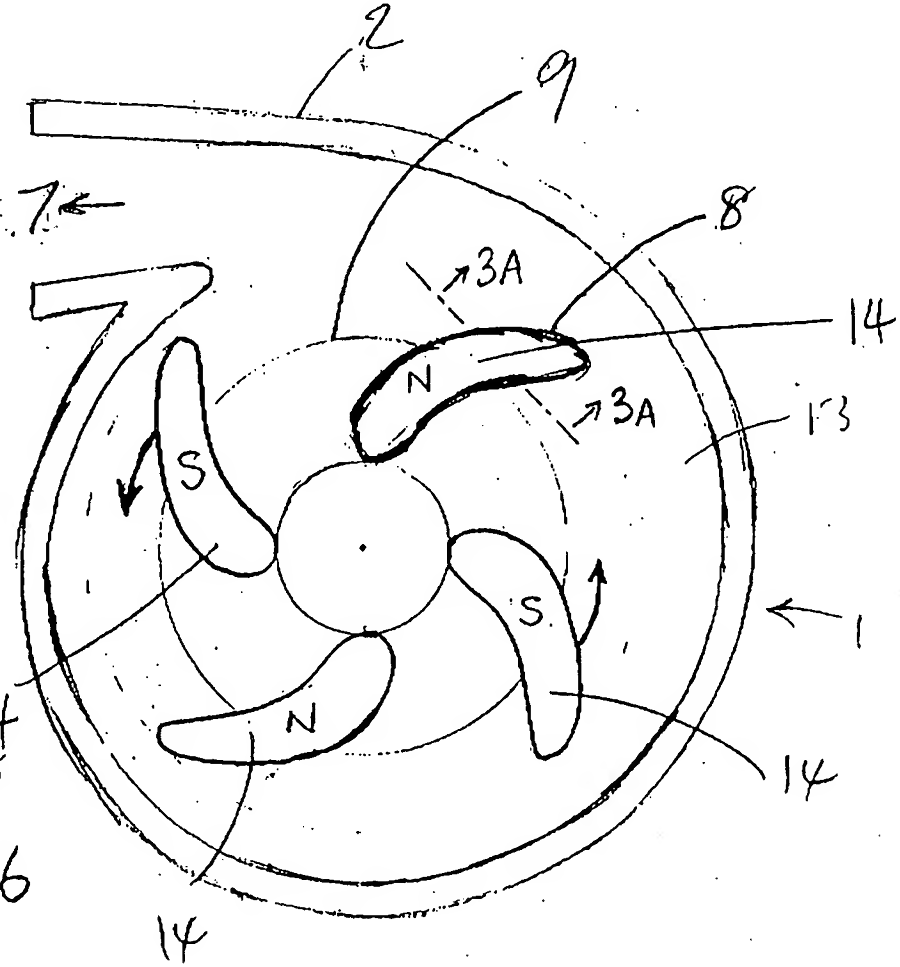

FIG. 2

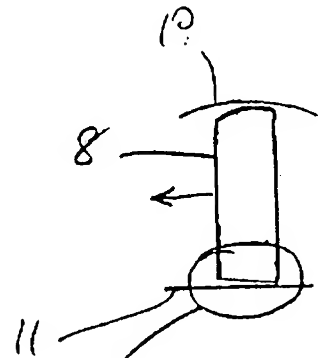

FIG. 3A

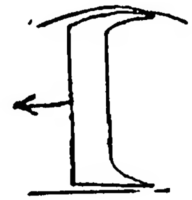

FIG. 3C

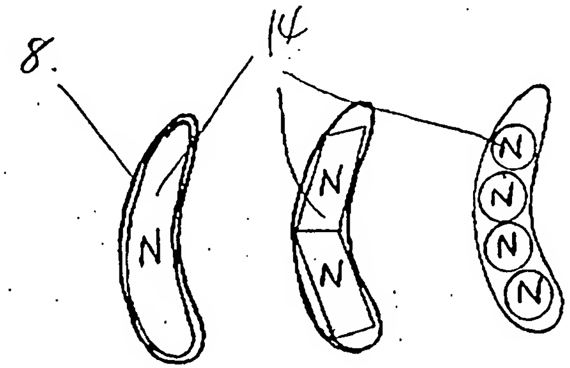

FIG. 4A 4B 4C

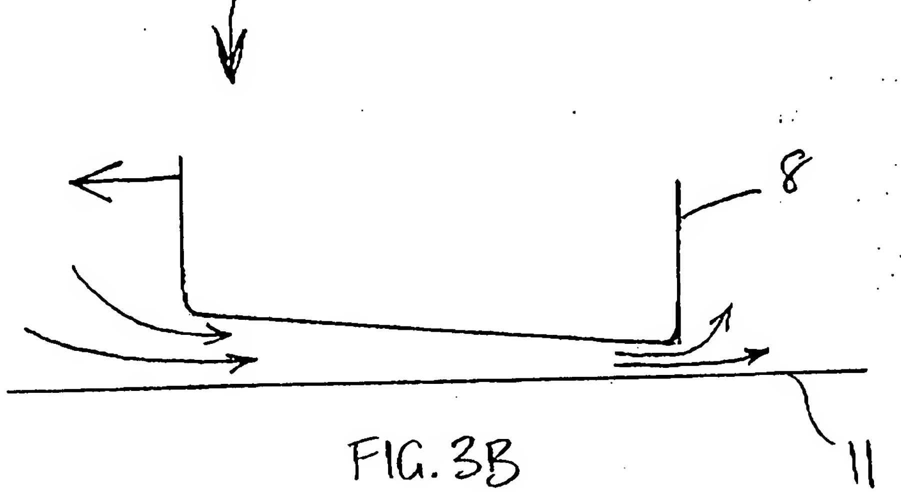

FIG. 3B

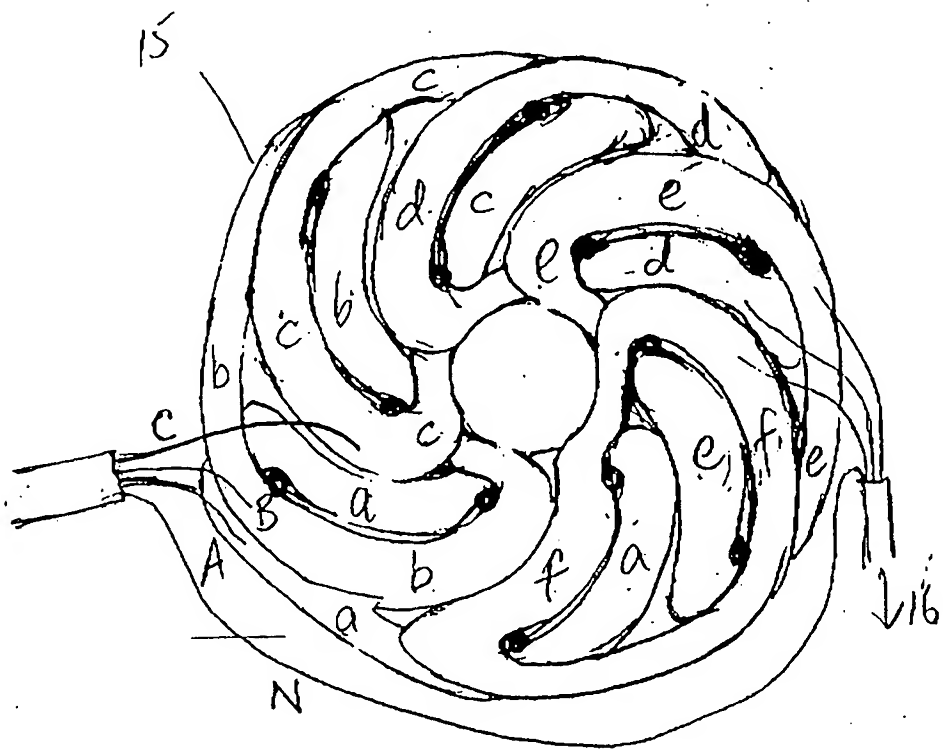

FIG. 5

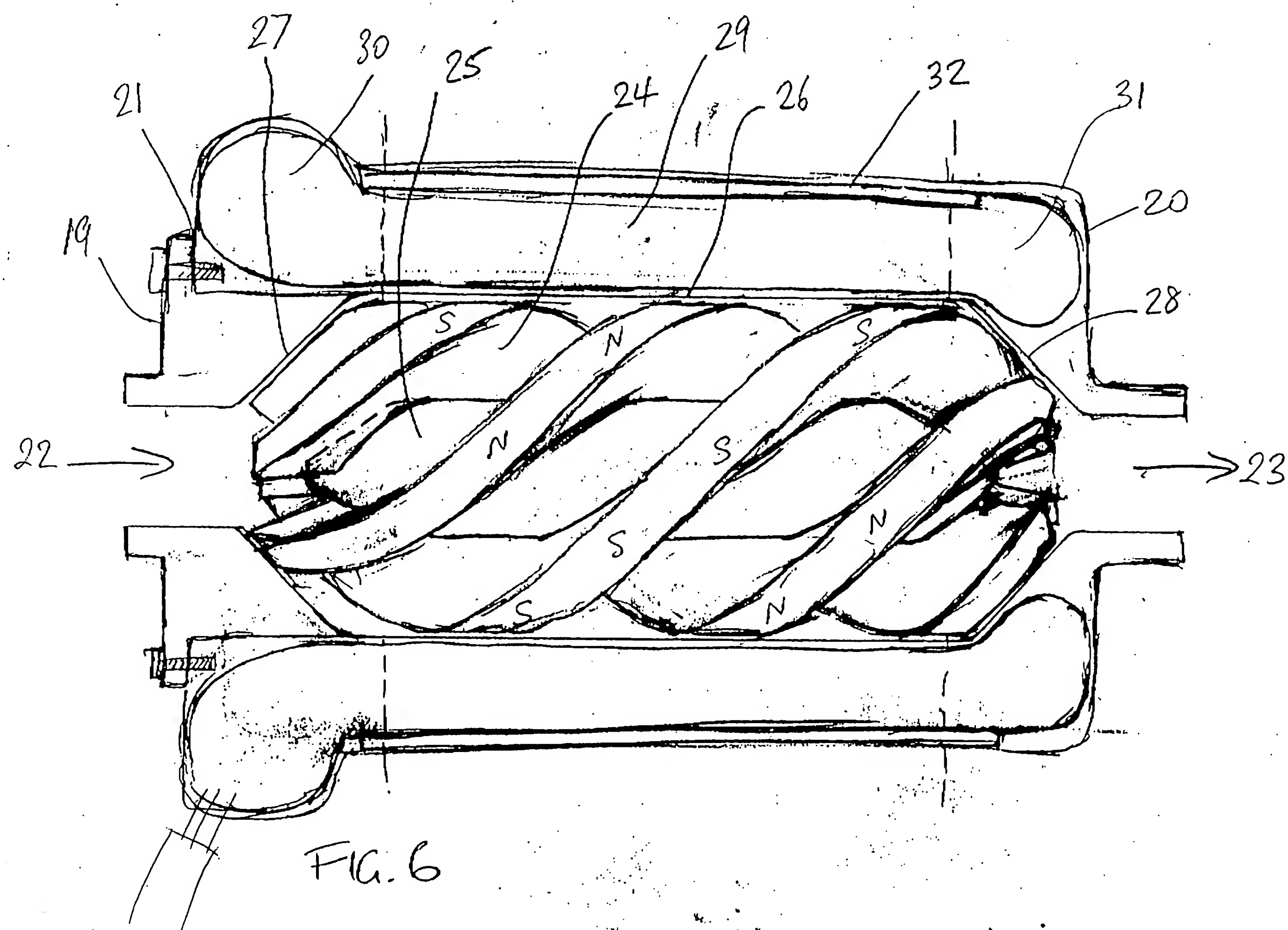

FIG. 6